RECENT EXPERIENCE WITH LARGE LIQUID INJECTED ROTARY SCREW PROCESS GAS COMPRESSORS

by

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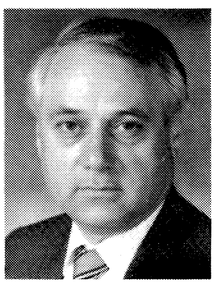

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ABSTRACT

Advances in manufacturing techniques and design enhancements for such components as shaft seals have made it possible for liquid flooded rotary screw compressors to achieve availability, reliability, and overall performance thought impossible 10 years ago. With some machines operating continuously for over 30000 hr, and installed driving power levels exceeding 7000 hp per compressor train, process engineers and machinery specialists may have to reassess older notions of the application range for this close cousin of the modern turbocompressor.

Specific details on construction features and design parameters are given for large rotary screw process machines which can clearly outperform turbocompressors and reciprocating process compressors in selected gas compression services. Two application examples are given, together with shortcut estimating procedures on the performance, initial installed cost, probable maintenance cost, and projected long term operating cost of large state-of-the-art rotary screw machines.

INTRODUCTION

Rotary screw compressors, sometimes called helical screw or spiral lobe compressors, belong to the rotary positive displacement machine category. For most users, they need little introduction or explanation; rotary screw compressors have been around for many decades and are very likely the "equipment of choice" for either oil-free or oil-wetted compression of air in mining, construction, industrial refrigeration, and a host of other applications where their relative simplicity, general reliability, and high availability are appreciated.

What is less well known is that rotary screw machines are equally suited to compress such process gases as ammonia, argon, ethylene, acetylene, butadiene, chlorine, hydrochloric gas, natural and synthetic pipeline gases, flare gas mixtures, blast furnace gas, swamp and biomass gases, coke oven or coal gas, carbon monoxide, town gas, methane, propane, propylene, flue gas, crude or raw gas, sulphur dioxide, nitrous oxide, vinyl chloride, styrene, and hydrogen. Modern sealing and liquid injection technology has been partly responsible for making rotary screw compressors capable of competing in applications previously reserved for other compressor types; however, there have been advances in other screw compressor component technologies as well. In the related areas of contour machining and metallurgy there has been additional progress. In any event, the mid 1970s sentiment expressed by contractors and at least one multinational petrochemical company limiting rotary screw compressors to relatively low discharge pressures and low power levels is worthy of critical review, update, and reassessment.

As of this writing, single or multistage rotary screw compressors cover the range of suction volumes from 300 to 60000 m^3/h (176 to 35310 acfm). Driver input values range as high as 5500 kW (7370 hp) and compressor discharge pressures of 40 bar (580 psi).

Examining both a typical and also a perhaps less conventional application for modern rotary screw machines is, thus, likely to represent a technology update for most readers. Moreover, comparing these rotary screw applications to their more traditional centrifugal and/or reciprocating equipment counterparts may prove especially valuable to engineers with review and selection responsibilities.

Dry vs *Liquid Injected Rotary Screw Compressors*

Modern process plants employ two slightly different types of rotary screw compressors: dry machines and wet, or liquid injected machines. Dry compressors typically use shaft-mounted gears to keep the two rotors in proper mesh. These machines found their earliest application in oil-free air services and are still prevalent in the pharmaceutical and laboratory grade chemical industries. It is certainly possible to envision compression and pneumatic conveying services for flour and sugar, or aeration services in the brewing industry, where the complete absence of oil carryover and other contaminants is deemed mandatory and which will, therefore, insist on the use of dry compressors.

Liquid injected rotary screw compressors do not usually require gearing to keep the two counter rotating screws in the proper mesh relationship. The injected liquid which could be water, a lubricating oil, or other liquid, provides a layer separating the two screw profiles even as one screw "drives" the other.

Liquid injected machines are often deemed highly advantageous because the user may benefit from one or more of their principal attributes:

- The injected liquid provides internal cooling. Certain gases are, thus, kept from polymerizing, or from operating in the explosion-prone temperature range.

- Liquid injected rotary screw compressors achieve considerably higher compression ratios than dry machines. It is entirely conceivable to have a single liquid injected compressor stage give the kind of pressure performance which would have mandated two or more of the "dry" compression stages.

Numerous compression services, thus, benefit from liquid injected compression processes. Several synthetic rubber plant recycle gases, ethylene oxide, acetylene, coal gases, and, in very special cases, even chlorine type gases, use liquid injected rotary screw compressors.

Background and Overview of Calculation Procedures

Since it is assumed that the reader is generally familiar with the overall configuration and principal operating characteristics of rotary screw compressors, only a few important principles will be highlighted.

A screw compressor (Figure 1) is a twin-shaft rotary piston machine functioning on the principle of positive displacement combined with internal compression.

Gas entering at the compressor suction flange is conveyed to the discharge port, entrapped in continuously diminishing spaces between the convolutions of the two helical rotors, being, thus, compressed up to the final pressure before it is expelled into the discharge nozzle.

The spaces referred to are those formed between the cylinder walls and the interlocking convolutions of the two helical rotors. The position of the edge of the outlet port determines the so-called "built in volumetric ratio" v_i. The "built in compression ratio," π_i, results from the equation, $\pi_i = v_i^x$, where x is the ratio of specific heats, otherwise known as the k-value.

The compression process is shown in the theoretical p-V diagram, Figure 2. This can be further visualized from Figure 3. A modern rotary screw compressor is designed for a desired or anticipated compression ratio π_i, as illustrated in Figure 2. If the

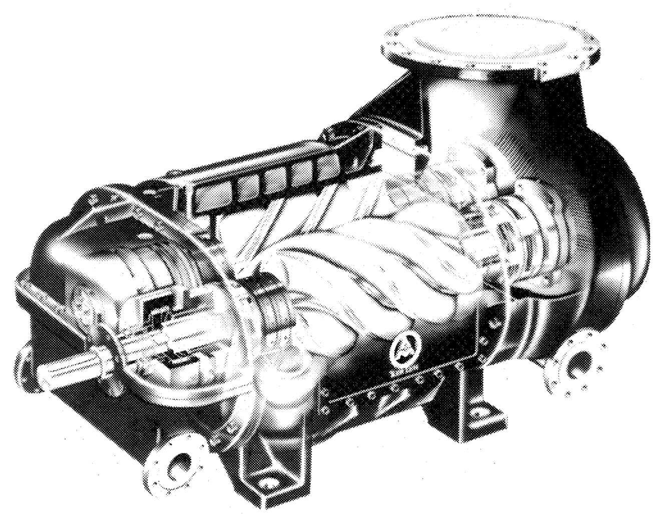


Figure 1. Twin-Shaft Rotary Screw Compressor.

compressor discharges into a receiver with a compression ratio in excess of π_i, the compressor end wall will be exposed to that pressure. Subject only to component strength and input power constraints, the compressor will be able to produce this increase in compression ratio or discharge pressure, whereas a centrifugal compressor would probably back up on its performance curve and encounter surge limitations. Less than designed-in compression ratios can be accommodated by the rotary screw compressor, also illustrated in Figure 2. In this case, some efficiency would be sacrificed.

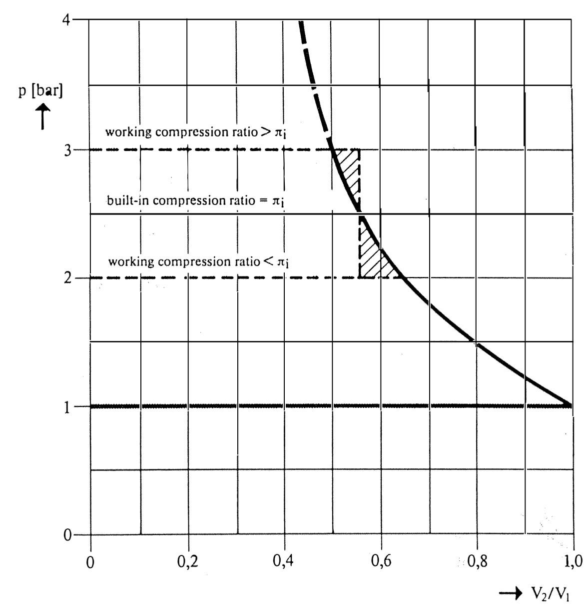
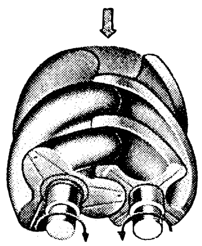


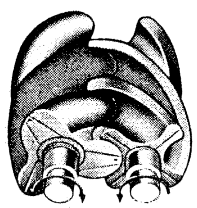
Figure 2. P-V Diagram of Rotary Screw Process Gas Compressor Operating under Different Discharge Pressure Conditions.

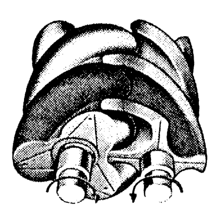
The induced flow volume may be calculated from any compression ratio, provided the data applicable to the particular compressor being considered are known. One revolution of the main helical rotor conveys the unit volume q_0(liter/rev).



Suction intake

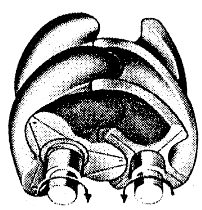
Gas enters through the intake aperture and flows into the helical grooves of the rotors which are open.





Compression process

As rotation of the rotors proceeds, the air intake aperture closes, the volume diminishes and pressure rises.



Discharge

The compression process is completed, the final pressure attained, the discharge commences.

Figure 3. Compression Sequence for Rotary Screw Compressor.

This gives us the theoretical induced flow volume Q_0 at n-revolutions:

$$Q_0 = \frac{n \cdot q_0}{1000} \ [m^3/min]$$

The actual induced flow volume Q_1 is lowered by the amount of gas Q_v flowing back through the very small clearances. Thus,

$$Q_1 = Q_0 - Q_v \ [m^3/min]$$

The slip loss volume Q_v is mainly dependent on the following individual factors:

- total cross-section of clearances
- density of medium handled
- compression ratio
- peripheral speed of rotor
- built in volumetric ratio.

Volumetric efficiency is expressed as

$$\eta_v = \frac{Q_1}{Q_0} = 1 - \frac{Q_v}{Q_0}$$

The theoretical power input required to compress the induced flow volume Q_1 is:

$$P_{th} = \frac{10^{-3}}{60} \cdot \delta_1 \, Q_0 \, h_{lad} \ [kW] ,$$

where δ_1 is the gas density at inlet conditions, kg/m^3,

and $h_{lad} \left[\frac{J}{kg} \right]$

represents the amount of energy required for the adiabatic compression of 1.0 kg of gas from p_1 to p_2.

Alternatively, the theoretical power input could be obtained from

$$P_{th} = \frac{k}{k-1} \left[\left(\frac{P_2}{P_1} \right)^{\frac{k-1}{k}} - 1 \right] \frac{Q_1 P_1 \times 10^4}{6000} \quad kW,$$

where $Q = m^3/min$ and $p_1 = bar_a$

The theoretical power input requirement is increased by the dynamic flow loss P_{dyn} and by the mechanical losses P_v. The latter consist of the losses in bearings, timing gears, and step-up gears.

Thus, the power transmitted through the coupling is:

$$P_k = P_{th} + P_{dyn} + P_v \ [kW]$$

North American screw compressors for important process applications are typically produced in compliance with API Standard 619. No negative tolerance is permitted on capacity and the power requirement may not exceed the quoted horsepower by more than four percent.

European screw compressors can easily comply with this requirement, although their customary specification (VDI 2045) would allow a different margin of deviation to accommodate tolerances resulting from the manufacturing process.

The final compression temperature is calculated for a dry-type compressor as follows:

$$t_{2\,th} = t_1 + \Delta t_{th} \ [°C] ,$$

where

$$\Delta t_{th} = T_1 \left[\frac{p_2}{p_1}^{\frac{x-1}{x}} - 1 \right] \frac{1}{\eta_v} \quad [°C]$$

When operating on the oil-free, dry-running principle, a screw compressor may come up to a maximum final compression temperature of 250°C (482°F).

When air is the medium handled, this temperature (isentropic exponent x = 1.4) corresponds to a compression ratio of

$$\frac{p_2}{p_1} \approx 4.5$$

On the other hand, gases with x = 1.2 will permit, within the temperature limits mentioned, a compression ratio of as high as

$$\frac{p_2}{p_1} \approx 7$$

In the case of a screw compressor operating on the principle of oil injection (Figure 4), most of the heat of compression is carried away by the oil. The amount of oil injected is adjusted to ensure that final discharge temperatures of approx. 90°C (194°F) are not exceeded. When taking in air under atmospheric pressure, compression ratios as high as $P_2/P_1 \sim 21$ are obtainable. For both butadiene and coke oven gas, ratios around 20 are equally feasible.

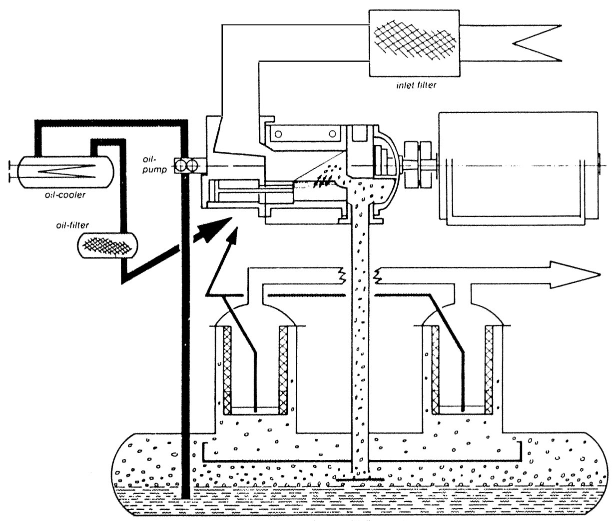


Figure 4. Principle of Liquid Separation Used with Liquid Injected Rotary Screw Compressors.

Advantages vs Disadvantages

As would be the case with virtually any other specific type or category of fluid machinery, rotary screw compressors embody advantages as well as disadvantages over other equipment competing for market share. Listing first the advantages, the application engineer would wish to consider the following:

• Considerably reduced sensitivity to molecular weight changes compared to centrifugal machines.

• Much greater tolerance for polymerizing service than other compressors, except perhaps liquid ring machines.

• Capable of accepting more liquid and fine solids entrainment than other compressors, except liquid ring compressors.

• Higher efficiency and less maintenance than liquid ring machines.

• Estimated availability in excess of 99.5 percent. This may approach or, in certain services, exceed that of centrifugal and axial compressors.

• Smaller size and lower cost than reciprocating compressors in same capacity range.

• Lower cost than centrifugal compressors in the small and moderate size ranges (below approximately 3000 kW (4000 hp).

• Higher pressure capability than other types of rotary positive displacement machines.

Among the disadvantages found are some that are perceived and others that are real. They, thus, merit more detailed examination.

• Sensitivity to discharge temperature which could affect close clearances and, hence, operability and availability: Proper temperature control instrumentation and generous sizing of cooling water or liquid injection facilities make this a non-issue for modern liquid injected screw compressors.

• Performance affected by rotor and casing corrosion/erosion. Increased clearances promote internal recycle or gas slip effects: Not a serious concern with water and oil-injected rotary screw compressors.

• Noise level is high enough to require silencing: A factor which must be taken into account. Capable rotary screw compressor manufacturers are fully equipped to provide well-engineered means of reducing environmental noise to meet even the most stringent requirements.

• Rotary screw compressor systems require pulsation suppression. While not as severe as piping pulsations encountered with equivalent reciprocating compressors, a properly engineered screw compressor system would incorporate appropriate pulsation bottles and, for high discharge temperatures, pipe expansion loops.

• Choice of rotor and casing materials more limited than for centrifugal compressors. This observation is related to the intricacies and close tolerance requirements of the machining process. Also, a knowledgeable manufacturer is cognizant of certain nonlinearities in the coefficients of expansion of different stainless steels. This might impose experience based temperature limitations on certain metallurgies and service conditions.

• Maintenance cost and duration of downtime higher than for centrifugals: Highly service dependent and not always so. Merits closer investigation.

• Flow control flexibility inferior to that of centrifugals and reciprocating compressors: A serious misconception which neglects to take into account the full spectrum of available options given later.

Capacity Control for Modern Rotary Screw Compressors

In principle, it is necessary to consider the problems of volume control for dry running and for oil injection-type screw compressors separately.

Volume control for dry screw compressors

Control by variable speed. Since screw compressors are positive displacement machines, the most advantageous method of achieving volume control is that obtained by variable speed. This may be done in any of the following ways:

· by variable speed electric motors
· by use of a torque converter
· by steam turbine drive

Speed may be reduced to about 50 percent of the maximum permissible speed. Induced flow volume and power transmitted through the coupling are in this manner reduced in approximately the same proportion.

Control by bypass. Using this method, the surplus gas volume is allowed to flow back to the intake side by way of a governor that is controlled by the allowable final pressure. An intermediate cooler reduces the temperature of the surplus gas volume down to the inlet temperature level.

Full load/idling speed governor. As soon as a predetermined final pressure is attained, a suitable transducer operates a diaphragm valve, which opens up a bypass between discharge and suction sides of the compressor. When this occurs, the compressor idles until pressure in the system drops to a predetermined minimum value. This will cause the transducer to initiate valve closure and the compressor will again be fully loaded.

Suction throttle control and discharge unloading. This method of control is suitable primarily for air compressors since temperature limits are quickly reached with this control mode. As in the case of the full load/idling speed control method, a predetermined maximum pressure in the system, for example, in a compressed air receiver, causes pressure on the discharge side to be relieved down to atmospheric pressure.

Simultaneously, the suction side of the system is throttled down to about 0.15 bar (2.2 psia) absolute pressure. When pressure in the whole of the system has dropped to a predetermined minimum value, full load is restored once again.

Control of Screw Compressors Equipped With Oil Injection

Suction throttle control. Since the final compression temperature is governed by the injected oil, it will be possible to operate over a wider range of compression ratios than would be feasible with dry machines. Suction throttling will, of course, increase the compression ratio of the machine and limitations will have to be observed. Nevertheless, oil injected machines can achieve stepless flow volume adjustment with relative ease.

Built in volume governor. Larger size compressors can be readily equipped with an internal volume regulating device (Figure 5). This device consists of a slide that is shaped to match the contours of the housing and which is built into the lower part of the housing. By moving this slide in a direction parallel to the rotors, the effective length of the rotors can be shortened.

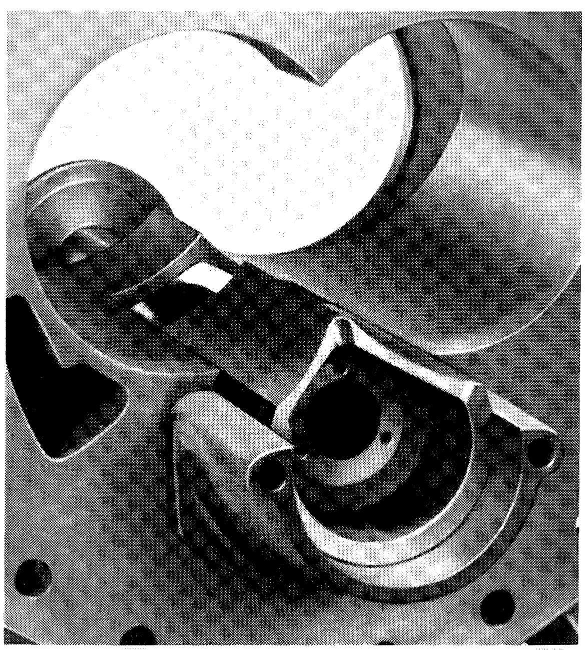


Figure 5. Slide Valve Used for Capacity Control of Rotary Screw Process Gas Compressor.

The range of stepless, infinitely variable control methods extends from 100 percent down to 10 percent of full compressor capacity. Slide control offers appreciable efficiency advantages over suction throttling.

Bearings, Lubrication, and Sealing

Rotary screw compressors, designed for high speeds and pressures, incorporate sleeve bearings and self adjusting multisegment thrust bearings, as illustrated in Figure 6.

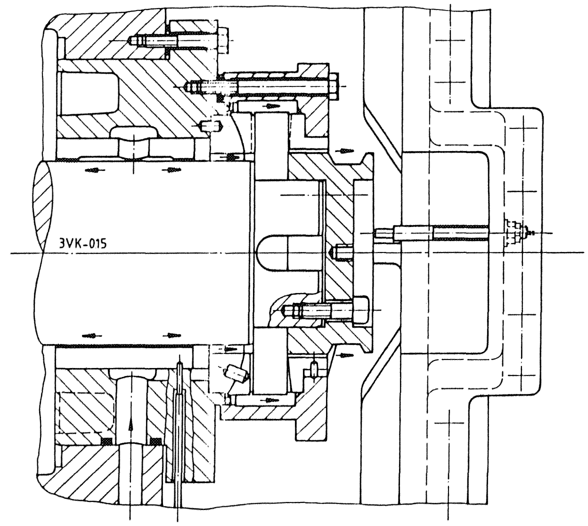


Figure 6. Self-Adjusting Multisegment Thrust Bearing.

Modern screw compressors can be equipped with the type of sealing system best suited for a particular process gas service and operating condition. Among these would be carbon ring seals (Figure 7), which are used in conjunction with buffer gas injection and leak-off ports connected back to compressor suction or flare gas collection headers.

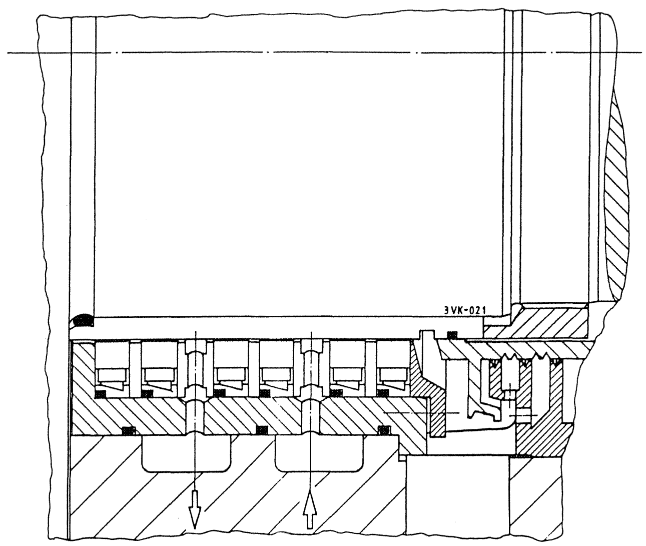


Figure 7. Carbon Ring Seals Used in Conjunction with Buffer Gas Injection.

Barrier water floating ring seals (Figure 8) allow a certain amount of water to reach the compression space. This water functions as a sealing, cooling, and flushing or gas scrubbing medium. Most of the barrier water is returned to the barrier water supply system for re-use. Stationary double mechanical seals, lubricated with pressurized water or a suitable oil (Figure 9), are successfully used in many other applications. Alternatively, a stationary single seal combined with a floating sleeve element (Figure 10) has served exceedingly well on machines with high differential pressures.

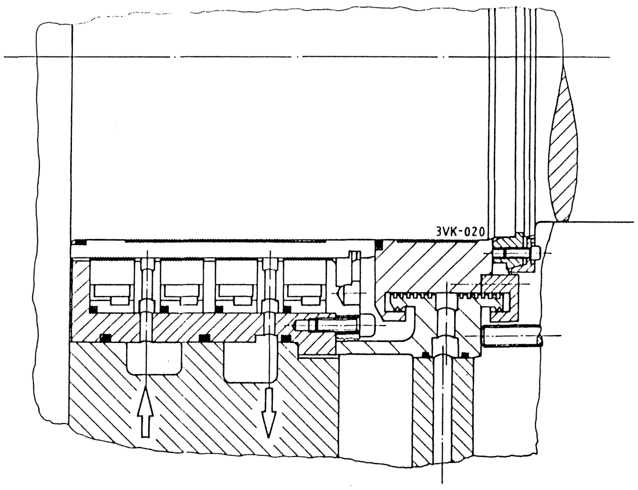


Figure 8. Barrier Water Floating Ring Seals Used Primarily on Liquid Injected Rotary Screw Compressors.

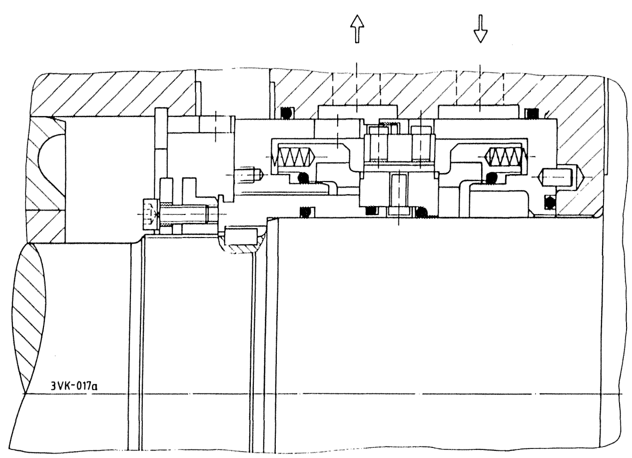


Figure 9. Stationary (Spring-Loaded Parts Nonrotating) Double Mechanical Seal for Rotary Screw Compressors.

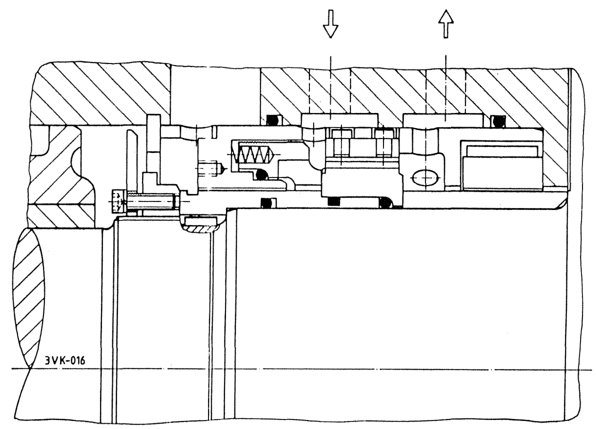


Figure 10. Stationary Single Mechanical Seal with Floating Sleeve Element.

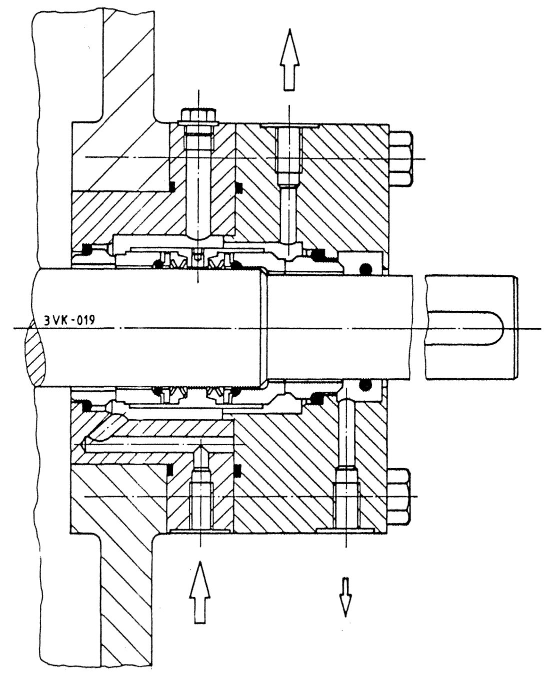


Figure 11. Traditional Mechanical Seal Used on Transmission Side of Rotary Screw Compressor.

Finally, the more traditional mechanical seals which are used on the transmission side of geared rotary screw compressor casings are illustrated in Figure 11.

Modern compressor components, new machining methods, and an awareness of the capabilities of advanced metallurgy have recently been brought together in thoroughly well engineered rotary screw compressors. Although quite close to traditional in outward appearance, these rotary screw compressors have consistently exceeded the size, reliability, availability, and maintenance cost constraints associated or perhaps perceived a few decades ago. The following examples will support this contention.

Application Example for Water Injected Rotary Screw Compressors In Coal Gasification Plants

Outstanding examples of the capabilities of modern two-stage, water injected rotary screw compressors can be found in several European coke gas producing plants. Derived from coal, this gas is utilized in steel making and as town blend-gas for heating and cooking. Previously installed centrifugal compressors proved highly vulnerable to performance degradation due to rapid polymerization of the hydrogen-rich, relatively dirty, gas. This necessitated the scheduling of costly downtime events for cleaning of compressor internals every four to six weeks.

One coal gasification plant replaced their centrifugal compressors with three two-stage rotary screw compressors (Figure 12), with water injection arranged as shown in Figure 13. Typical operating data are listed in Table 1, with and without booster for these three machines. The availability and reliability record for these 5.5 MW (7400 hp) electric motor driven water injected multistage rotary screw compressors is exceptionally good. Each of the three screw machines compresses approximately 33,000 Nm³/h (19420 cfm) of coke oven gas from one to 12 bar (14.5 to 174 psia).

These machines operate at considerably lower cost than the centrifugal compressors they replace. Both power and overall maintenance costs have been drastically reduced.

The decision to use screw compressors with water seals and with water being injected into the process stream was prompted by clear technical and thermodynamic considerations. With dry compression and discharge temperatures in excess of 100°C (212°F), higher hydrocarbons which tend to have a softening effect would have evaporated, leaving the asphalt-like components in the gas to coat both rotors and housings. Not only would these deposits adversely affect throughput and efficiency of other compressor types, but in the case of dry screw compressors, risk sticking of rotors if the machine had to be brought down and cooled for any reason.

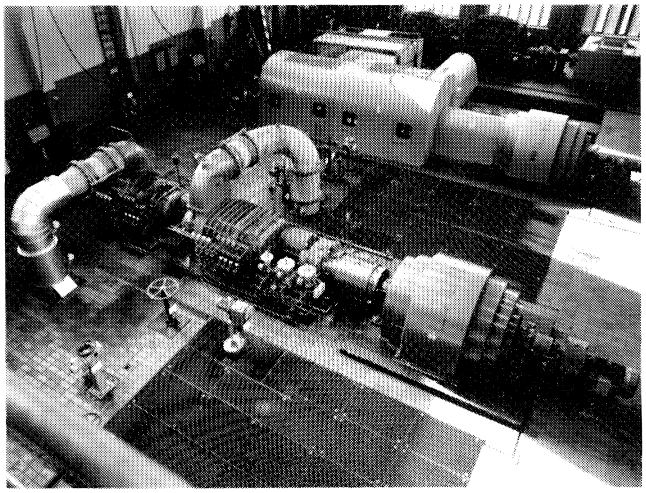


Figure 12. Two-Stage Water Injected Rotary Screw Compressors (5.5 MW, 7400 HP) in Coal Gas Service.

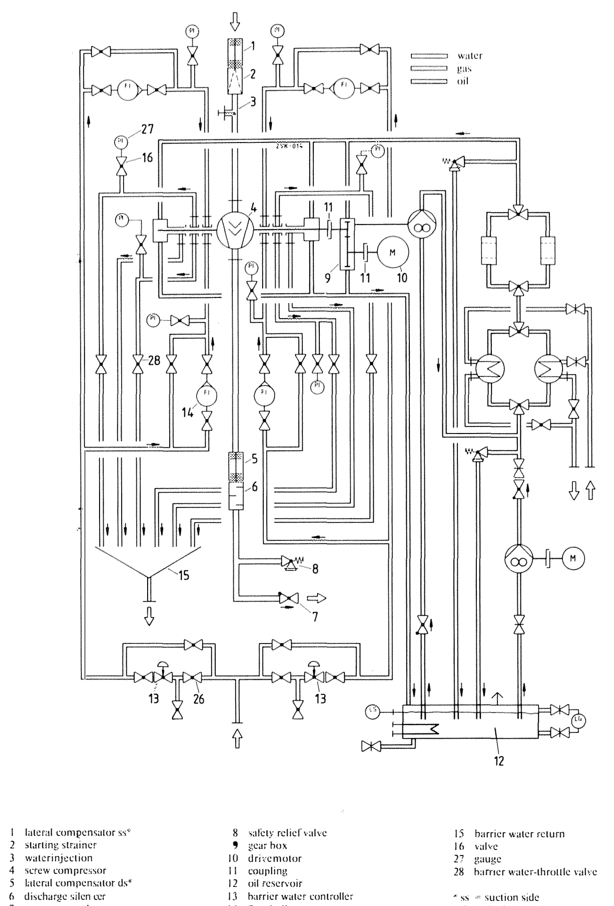


Figure 13. Flow Schematic Drawing of a Single-Stage Rotary Screw Compressor with Water Injection.

Table 1. Performance Chart For Two-Stage Water-Injected Rotary Screw Compressor Illustrated in Figure 12.

Compressor Stage		I	II
Inlet capacity	icfm	19,280	5,438
Inlet pressure	psia	23.9	75.4
Inlet temperature	°F	176	104
Discharge pressure	psia	76.9	174
Discharge temperature	°F	212	212
Compressor speed	rpm	3,250	3,250
Motor speed	rpm	1,500	
Power @ compr. shaft	bhp	6,566	
Motor rating	hp	7,370	
Lube oil pressure	psig	22 normal, 17.5 minimum	
Lube oil temperature	°F	120 normal, 160 maximum	
Oil capacity	Gallons	655	
Oil pump motor	hp	15	
Seal and injection water	gpm	42	
Seal water pressure	psig	174	
Injection water flow	gpm	5.7	10.5
Seal water into gas	gpm	8.75	8.75
Seal water return	gpm	24	
Water: for oil cooling	gpm	83	
Water: gas intercooling	gpm	864	

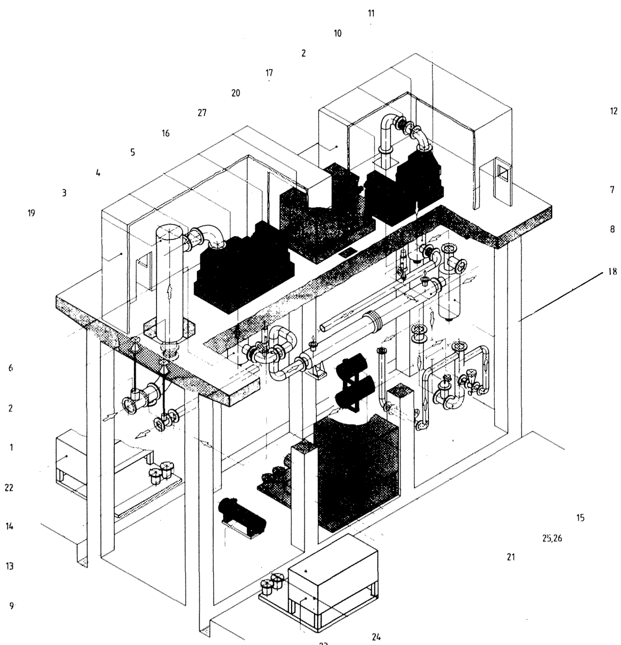
Liquid injection or, in this case, water injection, limits the final gas temperature to 100°C (212°F). Feeding a well defined amount of water into the compression space of the rotary screw compressor solves the polymerization problem and removes the heat of compression as the water proceeds to evaporate.

Most of the water is supplied through the four shaft seals of each stage (Figure 8). The remainder is sprayed into the inlet nozzle of each stage, with the injection rate controlled by a compressor discharge temperature transducer. The gas temperature is, thus, regulated to stay just below the dew point. Not only does this provide a cooling effect, but deposits are also flushed away by the excess water present. The water is drained off at the discharge silencer and in the water separators of the intercoolers and aftercoolers of each stage.

Because the gas contains corrosive components such as hydrogen sulphide, ammonia, hydrogen cyanide, and carbon dioxide, CrNi alloy steels have been used as compressor material. These steels are resistant not only to chemical attack by the gas, but also ensure the necessary erosion resistance. Corrosion related wear could occur as a result of the water injection.

An inspection was carried out after one year of continuous service in the coking plant with the three 7400 hp trains. It was quickly determined that rotors and housings were free of dirt deposits and showed no signs of damage due to corrosion or erosion.

At one plant (Figure 14), the volume flow of the screw compressors is matched to the momentary gas requirement by "intermediate pressure regulation." This means that the gas not required by the final receiver or downstream process is returned after the first stage, via a bypass to the intake side, so that the intermediate pressure drops. This power saving method provides a continuous range of regulation from approximately 20,000 Nm^3/h to 33,000 Nm^3/h (11770 to 19420 scfm). At 20,000 Nm^3/h (11770 scfm) the power consumption is only approximately 3200 kW (4290 hp) in comparison with 3950 kW (5295 hp) at 30,000 Nm^3/h (19420 scfm).



1 gate valve
2 lateral compensator
3 intake silencer
4 starting strainer
5 screw compressor 1st stage
6 discharge silencer 1st stage
7 intercooler
8 separator
9 safety relief valve 1st stage
10 starting strainer 2nd stage
11 screw compressor 2nd stage
12 discharge silencer 2nd stage
13 non-return valve
14 gate valve
15 control and shut-off devices
16 gear box 1st stage
17 gear box 2nd stage
18 safety relief valve 2nd stage
19 acoustic hood
20 acoustic hood
21 oil system
22 barrier water system
23 water injection system
24 condensate tank 1
25 condensate tank 2
26 condensate tank 3
27 drivemotor

Figure 14. Two-Stage Rotary Screw Compressor Unit in Coal Gas Service. Note separate systems for sealing water, injected water, and bearing lubrication.

This large gas producing plant has the additional capability of boosting the compressor inlet pressure to approximately 1.6 bar (23.2 psia) through the use of superchargers. This results in a compressor intake volume of approximated 46000 Nm^3/h (27070 scfm). Combining intermediate pressure regulation and supercharging, thus, provides a continuous range of regulation from 20,000 Nm^3/h to 46,000 Nm^3/h (11770 to 27070 scfm). With this wide tolerance range, the coking plant can adapt at any time to the momentary gas requirement.

The authors had access to operating data from a similar, smaller, two-stage water injected rotary screw compressor in coal gas service at yet another location. The performance is shown *(Appendix A)* at five different operating points, including 100 percent flow at different discharge pressures, 50 percent flow, minimum speed, and reduced pressure ratio.

Of course, applications of screw compressors are in no way limited just to coking plants. They can be used everywhere for delivery and compression of contaminated gases; moreover, they are ideally suited for gases which tend to polymerize at relatively low temperatures. Calcium kiln and pyrolysis gases belong in this category. However, there is also a wide range of possible applications in the chemical and petrochemical industries, as the next example will show.

Application Example for Rotary Screw Compressor In Butadiene Service

In the mid 1970s, a major US producer of ethylene specified a twice-intercooled centrifugal compressor for butadiene service. Operating conditions were as given in Table 2. Refer to *Appendix B* for calculated performance, *Appendix C* for graphical shortcut method (English units), or *Appendix D* for graphical shortcut method using metric units.

Table 2. Operating Data for Butadiene Compressor in the 1500 to 2000 Horsepower Category.

Gas Molecular Weight	54
$k = c_p / c_v$	1.14
Inlet Pressure P_1	1.3 psig = 16.0 psia = 1.1 bar
Inlet Temperature T_1	100°F = 560°R = 311°K
Flow Rate (m)	1170 lbs/min = 8.83 kg/s
Second Stage Discharge Pressure P_2	110 psia = 7.56 bar
Overall Pressure Ratio	P_2 / P_1 = 110/16 = 6.875

Since that time, this well-designed machine has performed flawlessly and there are no regrets. However, a replacement compressor alone at 1991 pricing levels would cost $1.5MM; with electric motor driver, speed increasing gear and lube/seal oil system the total estimated cost would be $2.3MM.

Confident that today's two-stage (once intercooled) rotary screw compressor would closely match this well performing centrifugal in such areas as power and maintenance cost (see APPENDICES for performance calculations) the screw machine would become a very strong and viable contender for this butadiene service. This is especially so since its "compressor only" cost is $1.2MM, and the total estimated cost, including gearing, electric motor driver, and lube/seal oil console is $2.0MM.

Maintenance Experience Is Favorable

The maintenance history of the three two-stage, 5.5 MW rotary screw compressors in coal gas service was reviewed (See Table 1 for performance data). This particular installation went through partial dismantling of the compressors every five years. At that time, only the carbon seal rings were being replaced. To date, all other parts, including bearings, have been reinstalled in the compressors without change, modification or correction. No problems have been reported and maintenance costs are estimated to be in the vicinity of $80000 per five-year period. This estimate includes labor and materials and is orders of magnitude below the expenditures incurred with centrifugal compressors in this particular fouling service.

A second installation operates two three-stage rotary screw compressors. This plant performed its first turnaround inspection after 35000 hr of uninterrupted service. Several carbon seal rings showed traces of wear, all other parts were found in excellent condition. However, since the carbon rings were still quite serviceable, this installation increased its projected turnaround intervals to 45000 to 50000 operating hours.

The third installation was less than perfect. This plant reported accelerated erosive wear of carbon seal rings, attributed to unsatisfactory water quality. Water injected rotary screw compressors require demineralized water and, in this regard at least, resemble the common steam turbine.

Summary: Don't Overlook Rotary Screw Compressors

Modern screw compressors are twin-shaft rotary piston machines; they operate on the principle of positive displacement combined with internal compression.

The performance of a screw compressor is influenced by factors such as gas properties, internal clearances, rotor length-to-diameter ratio, built in compression ratio, operating speed, compressor size, etc. Although compressor manufacturers have not found it practical to represent graphically the compressor performance for various machine sizes, gases, inlet and outlet conditions, rule-of-thumb estimates can be made with similar ease as for conventional centrifugal compressors. Typical adiabatic efficiency is almost always between 70 percent and 80 percent.

Modern rotary screw compressors can be mechanically loaded with pressure differences up to 12 bar, or 175 psi. Higher pressure differentials are possible for special cases.

The maximum allowable compression ratio for one screw compressor stage that will not cause the final compression temperature to rise above the permitted value of 250°C (482°F) will to a very large extent depend on the specific heat ratio c_p/c_v of the gas to be compressed. For example, where the specific heat ratio c_p/c_v is 1.4, the maximum compression ratio would be approximately 4.5, and where the specific heat ratio c_p/c_v is 1.2, the maximum compression ratio would be approximately 10.

With multistage arrangements, final pressures of 25 bar to a maximum of 40 bar (362—580 psig) can be achieved. For vacuum applications, an absolute pressure of 0.9 bar (13 psia) can be reached. The process gas screw compressors are equipped with intermediate cooling.

Compressor speeds vary from about 2000 rpm to 20000 rpm, depending on compressor size. The peripheral speed of the rotor governs the rpm of the male rotor. This peripheral speed ranges from 40 to 120 m/s (131—394 fps) up to maximum of 150 m/s (492 fps) for gases with a low molecular weight.

Finally, oil or water-flooded machines open up an entire new range of applications for numerous process gas streams. These machines make it possible to avoid polymerization by keeping the compressor discharge temperature below the dew point. This permits single staging where other machinery types might require multistaging. The resulting economies can be substantial.

APPENDICES

Appendix A

The operating data tabulated below reflect the remarkable flexibility of a two-stage water-injected rotary screw compressor in coal gasification or coke oven service.

Appendix A.

STAGE 1	STAGE 2
$P_2/P_1 = \pi = 3.5$	$P_2/P_1 = \pi = 3.0$
L/D = 1.794	L/D = 1.14
T_l = 300°K = 80°F	T = 300°K = 80°F
q_0 = 133.63 liters/rev	q_0 = 22.95 liters/rev

OPERATING POINTS

	I	II	III	IV	V
P_1 (stage 1/2) bara	1.06/4.5	1.06/4.3	1.06/4.4	1.06/4.9	1.06/4.2
P_1 (stage 1.2) psia	15.4/65.3	15.4/62.3	15.4/63.8	15.4/71.1	15.4/60.9
P_2 (stage 1/2) bara	4.6/13.7	4.4/9.0	4.5/9.0	5.0/13.7	4.3/5.0
P_2 (stage 1/2) psia	66.7/198.6	63.8/130.5	65.3/130.5	72.5/198.6	62.3/72.5
t_1 stage 1/2) °C	30/35	30/35	30/35	30/35	30/35
t_1 (stage 1/2) °F	86/95	86/95	86/95	86/95	86/95
t_2 (stage 1/2) °C	82/95	80/76	83/77	92/103	80/58
t_2 (stage 1/2) °F	180/203	176/169	181/171	198/217	176/136
Q_n (nm³/h//cfm	28270/16637	28270/16637	14135/8318	9080/5344	28550/16802
P_d (kw/hp)	3894/5220	3248/4342	1815/2433	1686/2260	2866/3842
Remarks	100 % Q_1	100 % Q_1	50 % Q_1	min (RPM)	5 bara

GAS COMPOSITION

Volume %		Grams/m³	
CO_2	0.7 – 1.3	H_2S	0.5 – 1.7
Heavy Fract.	3.6 – 4.0	NH_3	0.02 – 0.05
O_2	0.1 – 0.2	HCN	0.5 – 1.0
CO	4.6 – 6.1	C_6H_6	35 – 45
H_2	54.9 – 57.8	Org. S	0.5
CH_4	26.2 – 27.7	Tar	10 – 20 mg/m³
N_2	6.3 – 6.4		

100 % H O - saturated

Barrier water sealing loop: 8.5 m³/h, with 3.0 m³/h going into the machine and 5.5 m³/h being returned to the supply system.

Appendix B. Gas Conditions for Butadiene Compressor

Butadiene: MW 54 , $k = C_p/C_v = 1.14$
Inlet Pressure = 1.3 psig = 16.0 psia
Inlet Temperature = 100°F = 560°R = 311°K
Flow Rate (m) = 1170 lbs/min = 8.83 kg/s

$$PV = mRT,\ V = \frac{mRT}{P} = \frac{(1170)\ (154)\ (560)}{(144)\ (16)\ (54)}$$
$$= 8131 \text{ cfm (or from chart)}$$
$$= 3.84 \text{ m}^3/\text{s}$$

Second Stage Discharge Pressure 110 psia

Overall Pressure Ratio: $P_2/P_1 = \pi = \frac{110}{16} = 6.875$

Per-Stage Pressure Ratio : $(6.875)^{.5} = 2.63$

Head Per Stage*

$$\frac{n}{n-1} = \left(\frac{k}{k-1}\right)\eta_p \left(\frac{1.14}{1.14-1}\right) .77 = 6.27$$

$$H = ZRT\left(\frac{n}{n-1}\right)\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$$= \left(\frac{1545}{54}\right)(560)\ (627)\ \left[(2.63)^{.16} - 1\right]$$

$$= 16800 \text{ ft}$$

Horsepower*

$$hp = \frac{\text{(Weight Flow) (Head)}}{(3300)\ (\eta)} = \frac{\dot{m}H}{(33000)(0.75)}$$

$$= \frac{(1170)\ (16800)}{(33000)\ (0.75)} = 794 \text{ hp/stage}$$

For both stages: ~ 1588 hp + ~ 100 hp for bearings and seals
= ~ 1688 hp

Discharge Temperature*

$$T_0 = T_2 \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

$$= 560\ (2.36)^{.123} = 631°R = 171°F$$

*All calculations approximate and intended for scoping studies only

Appendix C

A graphical quick selection method for multistage centrifugal compressors was developed in the mid 1970s by Don Hallock, an engineering manager with the Elliott Company, Jeannette, Pennsylvania.

Hallock was able to reduce this quick selection method to a series of charts (Figures C-1 through C-6). To use these charts, the following quantities must be known:

Appendix C

W	Weight flow in lbs. per min., of scfm - standard cu. ft. per min.
P_1	inlet pressure in psia
R_p	pressure ratio (discharge psia/inlet psia)
t_1	inlet temp., °F
M	mole. weight
K	ratio of specific heats

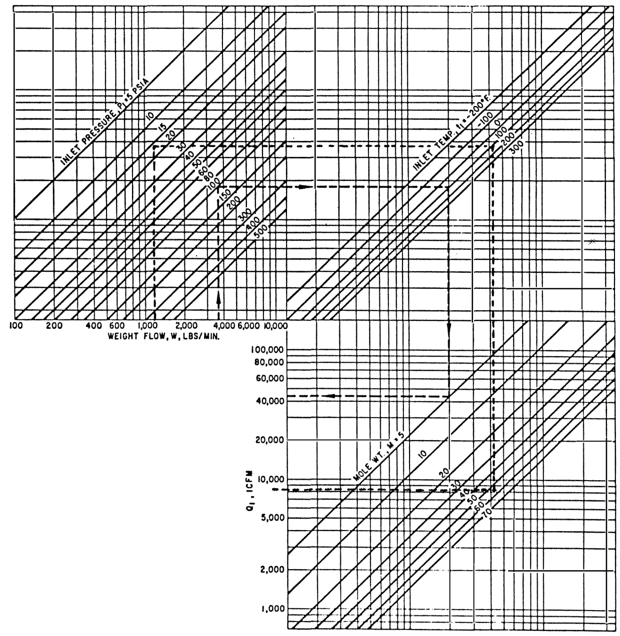


Figure C-1. With Weight Flow of Butadiene Gas Known (1170 lb/min), enter Chart at W, Draw Vertical Line to Inlet Pressure (16 PSIA), Go Horizontally to Inlet Temperature (100°F), Draw Vertical Downward Line to Molecular Weight (54), Read Inlet Volume Flow Q = 8130 ACFM.

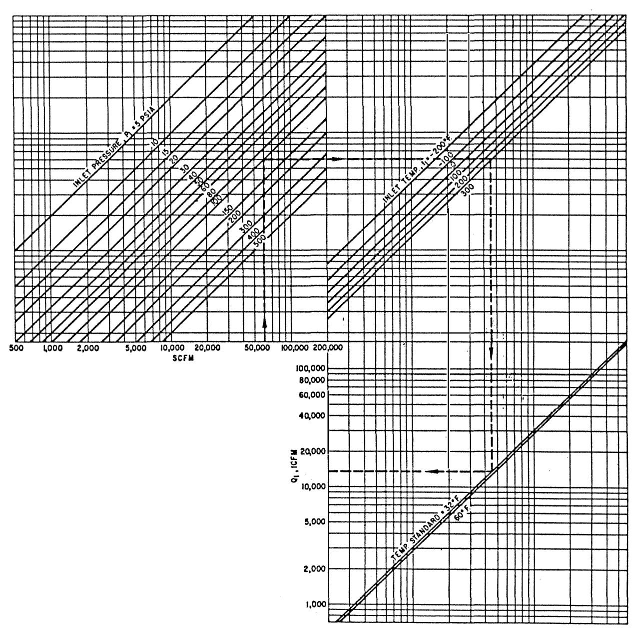


Figure C-2. If SCFM Is Known, Proceed through P_1, t_1, and "Temperature Standard" Usually 60°) to Find Q_1.

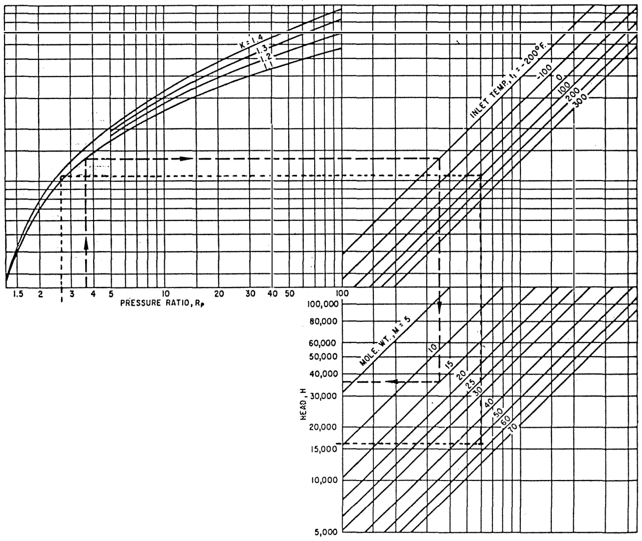


Figure C-3. To Find Head, H, Enter This Chart at Pressure Ratio R_p (= P_1/P_2 in PSIA), and Go through Inlet Temperature and Molecular Weight. (This is done on a per-process-stage basis). In our example, R_p = 2.63.

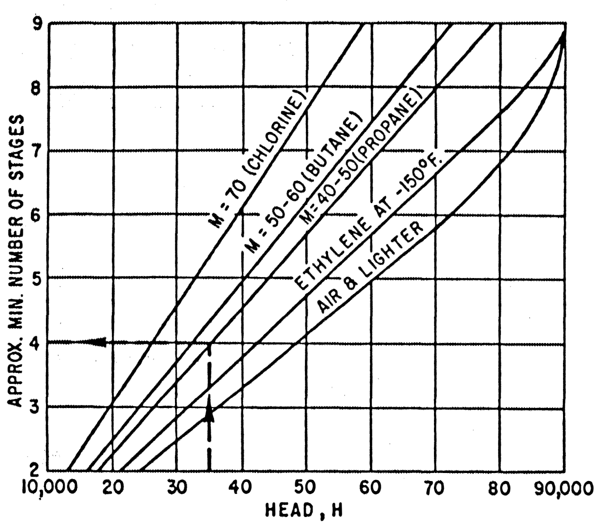


Figure C-4. Entering This Chart at a Given Head Allows Us to Read Off the Required Number of Stages (Impellers). Heads in excess of 80000—100,000 ft usually require more than one Centrifugal Compressor Casing.

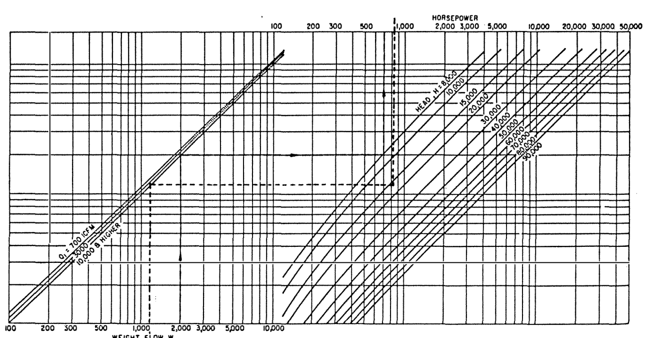


Figure C-5. The Horsepower Requirement Is Determined from This Chart. Enter W, proceed through Q_1 and H, read off hp (in this example case, about 820 hp per compressor section).

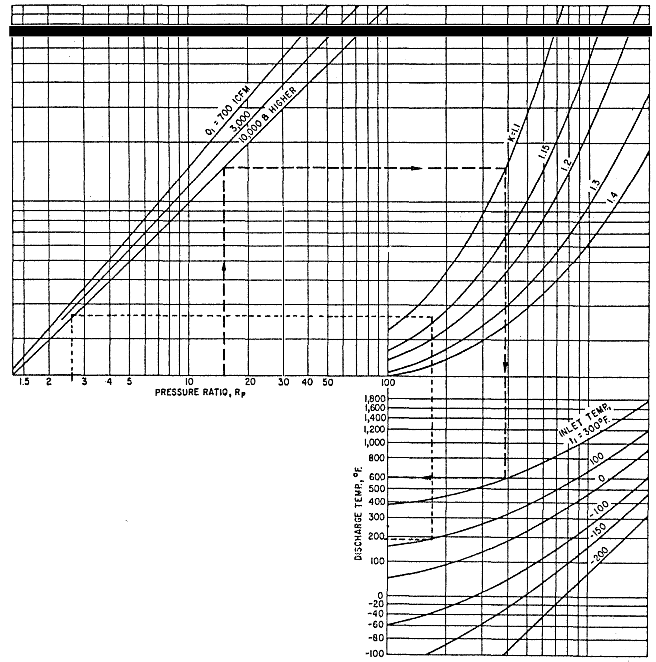


Figure C-6. The Discharge Temperature is Determined by Entering the Pressure Ratio R_p, Proceeding through Q_1 and K to Inlet Temperature t_1.

- To determine inlet cfm, Q1, use Figure C-1, if W is known.
- If scfm is known, use Figure C-2 to find Q1.
- To find head H, use Figure C-3.
- The number of stages (impellers) can be obtained from Figure C-4.
- Horsepower is determined from Figure C-5.
- The compressor discharge temperature can be determined from Figure C-6.

Appendix D

Figures D-1 through D-3 are representative of a graphical quick selection method for multistage centrifugal compressors developed by Mannesmann-Demag in the mid 1980s. The basic performance of centrifugal compressors can be approximated from a diagram which incorporates power, number of stages, speed, etc.

These diagrams are valid for isentropic, i.e., uncooled compression. A typical compressor efficiency has been "designed into" the respective diagrams. Compressors with pressure ratios up to 7:1 are covered by these graphical representations, since intercooling is almost always required for higher pressure ratios. In order to determine the compressor data in such cases, the compression process is split at the point where intermediate cooling takes place.

Using the example of the butadiene compressor where polymerization is to be avoided, the plan is to use intercooling after the first section. The second section is treated separately or, in this case, identically and the results added to approximate the total absorbed power for this compressor.

Calculation Example

For the butadiene compressor, approximate operating data as given below are sufficiently accurate for scoping purposes:

Appendix D

Molar mass	M	= 54
Compressibility	Z	= 1.0
Isentropic exponent	k	= c_p/c_v = 1.14
Mass flow rate	$\dot{m}$	= 31800 kg/h = 1170 lbs/min
Intake pressure	p	= 1.1 bar = 16 psia
Intake temperature	T	= 311°K = 100°F
Discharge pressure	p	= 7.6 bar = 110 psia
Max. tip speed (assumed)	u_2	= 240 m/s = 787 fps

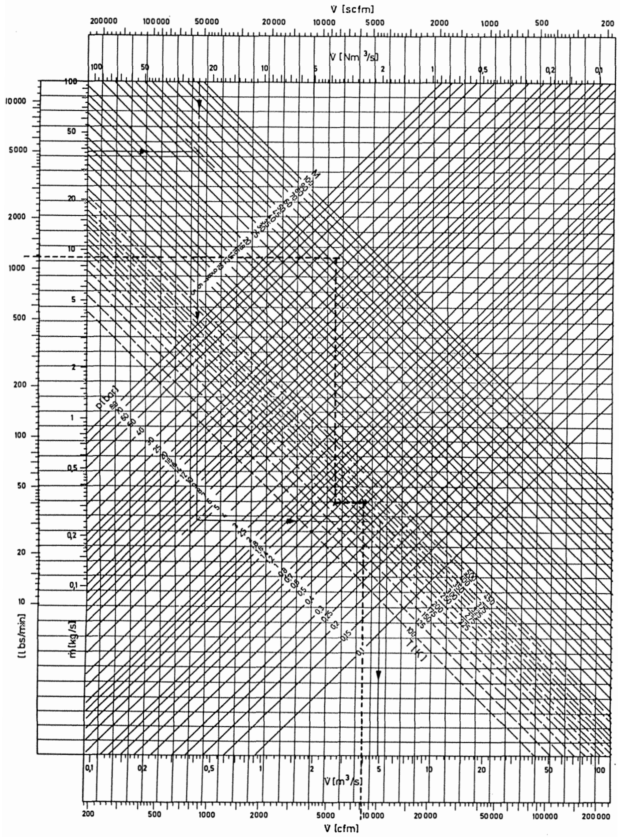


Figure D-1. Graphical Approximation Method for Determining Speed, Power Demand, and Number of Centrifugal Compressor Stages (Part 1).

The scoping procedure is as follows:

• From the intake pressure p_1 and discharge pressure p_2 the following pressure ratio is obtained:

$$\pi = p_2/p_1 = 7.6/1.1, \text{ or } 110/16 = 6.875$$

Pressure ratio for each of the two sections:

$$(6.875)^{0.5} = 2.63$$

• Correction of the ideal molar mass M using the real gas factor Z:

$$M = M_{id}/Z = 54/1 = 54$$

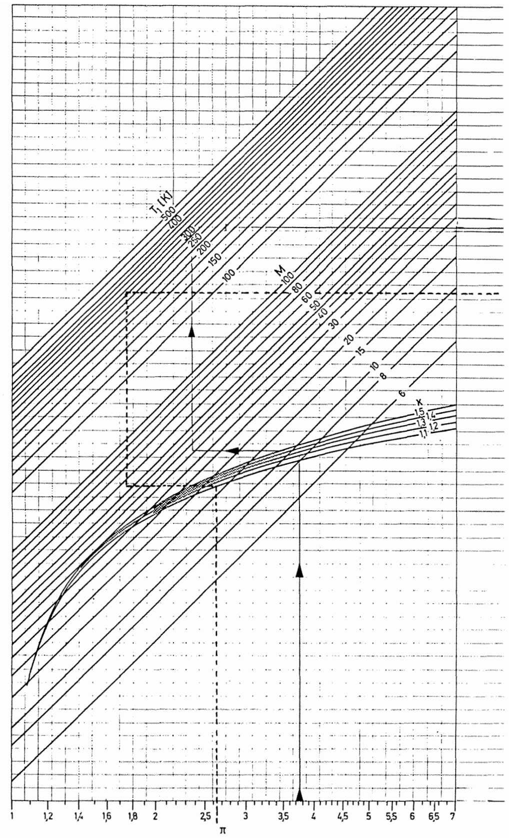


Figure D-2. Graphical Approximation Method for Determining Speed, Power Demand, and Number of Centrifugal Compressor Stages (Part 2).

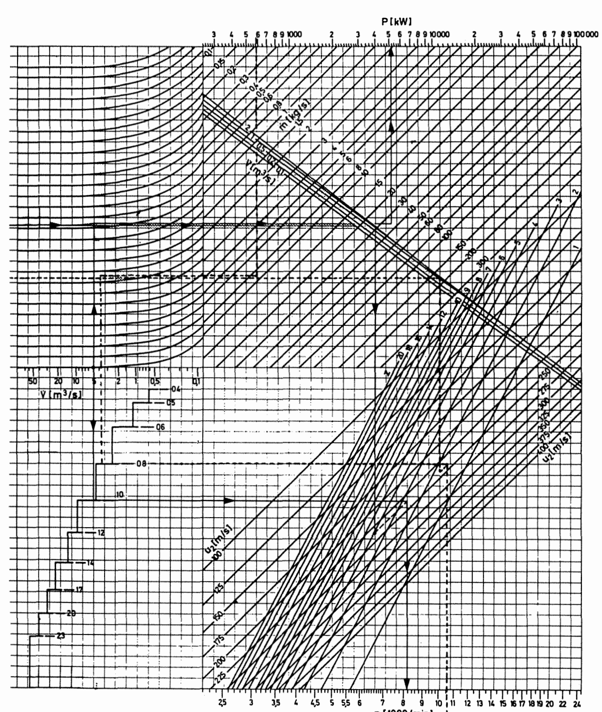


Figure D-3. Graphical Approximation Method for Determining Speed, Power Demand, and Number of Centrifugal Compressor Stages (Part 3).

• The data covering the volume flowrate $\dot{V}$ and the mass flowrate $\dot{m}$ for any quantity can be taken from the auxiliary diagram, Figure D-1, for the quantity relationship at given operating conditions. Taking M, p, and T into consideration, the following volume flowrate is obtained:

$$v = 3.84 \text{ m}^3/\text{s}$$

• Next, refer to Figure D-2. Starting from pressure ratio π = 2.63, the arrow points vertically upwards to the point of intersection, with the curve for the isentropic k = 1.14; from there the next arrow leads horizontally towards the point of intersection with the curve for the same molar mass M = 54.

Vertically above that point, the temperature line T_1 = 311°K is reached, after which the next arrow points horizontally towards Figure D-3 and the range of influence of the volume flowrate $\dot{V}$. Continuing on Figure D-3, the line then runs parallel to the family of curves until the specified volume flowrate $\dot{V}$ = 3.84 m^3/s is reached, after which the line runs horizontally again until it intersects with the corresponding mass flowrate line $\dot{m}$ = 8.83 kg/s. Vertically above this point of intersection, the power requirement P = 590 kW can be read off.

• The line is now extended horizontally through the range of influence of the volume flowrate $\dot{V}$ until the line reaches the volume-dependent lines of opposite inclination.

From this point, it is extended vertically downward as far as the stage number line z. For the assumed tip speed of u_2 = 240 (787 fps) m/s, then obtain z = 2 as the number of stages.

• Starting from the volume flowrate scale $\dot{V}$ = 3.84 m^3/s the line in the direction of the arrow then impinges vertically on a specific size of compressor casing. From that point the line is taken horizontally to the right until it intersects the line u_2 = 240 m/s (787 fps) established above. Vertically beneath this point of intersection, the speed n = 10500 rpm is read off.